



Energy system feasibility study of an Otto cycle/Stirling cycle hybrid automotive engine

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ABSTRACT

The aim of this study was to investigate the feasibility of utilising a Stirling cycle engine as an exhaust gas waste heat recovery device for an Otto cycle internal combustion engine (ICE) in the context of an automotive power plant. The hybrid arrangement would produce increased brake power output for a given fuel consumption rate when compared to an ICE alone. The study was dealt with from an energy system perspective with design practicalities such as power train integration, location of auxiliaries, manufacture costs and other general plant design considerations neglected. The study necessitated work in two distinct areas: experimental assessment of the performance characteristics of an existing automotive Otto cycle ICE and mathematical modelling of the Stirling cycle engine based on the output parameters of the ICE. It was subsequently found to be feasible in principle to generate approximately further 30% useful power in addition to that created by the ICE by using a Stirling cycle engine to capture waste heat expelled from the ICE exhaust gases over the complete range of engine operating speeds.

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1. Introduction

The idea of utilising a Stirling cycle engine as a waste heat recovery device on an automotive engine has received attention on numerous occasions to date. At least two major international automotive manufacturers have developed the idea in the past with slightly different applications in the context of automotive power plants [1–3]. Though the Stirling cycle engine has traditionally struggled to gain acceptance as a prime mover in its own right in competition with the Otto cycle engine, its use in the manner presently proposed, as a complimentary technology, may facilitate its acceptance somewhat and promote more diverse use of it as a clean, quiet alternative to some existing power plants. It has benefited from significant recent development [4–10], particularly in terms of distributed generation applications [11–13]. The primary issue for the present study, though, is improvement of the traditional automotive plant efficiency. This paper details a preliminary study undertaken to assess the viability of an automotive hybrid engine based on the combination of the Otto and Stirling cycle engines.

2. Methodology

The study involved two distinct areas of investigation. First, experimental data pertaining to the Otto cycle engine were collected and analysed. Second, a mathematical model of a potentially suitable Stirling cycle engine was generated in accordance with published methods. This enabled an assessment of the feasibility of such a coupling in terms of the energy system involved.

3. Experimental assessment of Otto cycle engine output parameters

The study required assessment of certain outputs of the Otto cycle engine, namely brake power, brake torque, engine speed, exhaust gas temperature and exhaust gas energy content.

The power and torque were measured using a rolling road dynamometer facility. This allowed measurement of road power, with brake power consequently calculated through the system software by allowing for transmission drag in the system.

Exhaust temperature was measured using a K-type thermocouple placed into the exhaust stream behind the manifold and read using a thermocouple reader.

As it was not possible to incorporate a calorimeter on the exhaust gas stream, exhaust energy was estimated in terms of a first law energy balance applied to the system in accordance with known operational data for Otto cycle engines. The energy balance of the system is analysed in Section 3.1.

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3.1. Internal combustion engine energy balance

The energy balance of a spark ignition (SI) internal combustion engine (ICE) such as the Otto engine can be represented by the equation [14]:

$$m_f h_f + m_a h_a = P_b + Q_{cool} + Q_{misc} + (m_f + m_a) h_e \quad (1)$$

where m_f and m_a are the fuel and air mass flow rates respectively, h_f and h_a are the fuel and air enthalpies, P_b is the brake power, Q_{cool} is the portion of energy transferred to the engine coolant, Q_{misc} is representative of the energy lost to the lubrication oil, and as convection and radiation from the engine's external surfaces and h_e is the exhaust gas enthalpy. This is an elementary first law balance that provides valuable insight into the operation of the engine. However, as ICE's are so versatile a device and operate in widely varying loading environments, it is difficult to offer a meaningful generalisation of the typical energy balance of such an engine in percentage terms. Fig. 1 offers some information on such a balance for a range of industrial SI engines [15]. Presented are the proportions of the three main energy sinks for a number of discrete engines; brake power, exhaust gas and jacket cooling water. The engine models are from the same manufacturer and are presented in order of increasing rated brake power output. From this we can see that at full load conditions and fixed speed (1500 rpm), these units convert between 37% and 43% of the available energy to brake power; lose 17%–28% to the exhaust and approximately 15%–34% to the jacket cooling water. From this it is notable that across the range there is significant difference in the proportions given to each sink, making it difficult to offer a definitive energy balance for the Otto cycle engine on the whole. From this a reasonable insight into the varying nature of the Otto cycle heat balance can be acquired. These variations may be due to differences in design details in the individual engine models, such as fuel delivery systems, ignition control and compression ratio. In automotive applications, generalisation is made more problematic with the added variables of a wide operational speed regime and fluctuating load environments. For the purpose of this study, it was therefore necessary to formulate an approximate energy balance for the automotive engine under examination.

Table 1 offers an approximation for the energy balance of the spark ignition (SI) automotive ICE under full load conditions [14,16,17].

The heat rejected from such an engine is a function of a number of variables – speed, load, ignition timing, fuel mixture, compression ratio for example. As so many dependent variables are

Table 1

Approximate energy balance of automotive spark ignition ICE.

Chemical energy of fuel	100%
Of which	
Energy lost to coolant	10%
Energy to brake power	28%
Energy lost to exhaust	25%
Energy lost to incomplete combustion of fuel	2%
Energy lost to radiation	35%

involved, it is difficult to generalise the proportions. The above percentage breakdown is considered a reasonable estimate of such an engine over the full speed range and full load.

3.2. Testing of ICE

In order to assess the feasibility of the proposed hybrid engine, it was necessary to ascertain the output parameters of a spark ignition ICE under typical operating conditions. This was done using a Renault Safrane 2.0 16V. The testing was done using a MAHA LPS 2000 Rolling Road Dynamometer system [18]. The rpm, engine oil temperature, ambient temperature, torque and power were monitored through the system software. Power was measured as road power, with the brake power calculated from measurement of the transmission drag during the test. The experimental procedure used is detailed below.

3.3. Test preparation

The car was set onto the rollers, a set of four rotating drums recessed into the laboratory floor. The rollers are connected to the dynamometer that is connected to an analogue/digital data acquisition system and PC. This was used to record the torque output at the driving wheels and subsequently the road power. An inductive pickup was attached to the top of the cylinder block for recording of rpm and the corresponding cylinder number was recorded. The engine oil temperature was monitored by placing a temperature probe into the sump via the dipstick hole. Monitoring of oil temperature during the test was used as an indicator for the engine operating temperature and allowed assessment of engine overheating. The engine was to maintain an operating oil temperature between 358 K (85 Celsius) and 373 K (100 Celsius). If the engine exceeded 373 K it was considered to be overheating and had thus exceeded its safe operating limit.

Exhaust gas temperatures were recorded by placing a K-type thermocouple into the exhaust stream immediately after the manifold and recording the readings using a thermocouple reader. A K-type thermocouple was used because of its suitability for use in electrically noisy environments such as those encountered in automotive engines. The thermocouple was mounted in the exhaust pipe immediately after the manifold. It was secured in place in a tapping point using exhaust repair putty and exhaust bandaging. Ambient temperature and pressure were recorded using a separate sensor. This was necessary for setting of the DIN 70020 standard for the engine test. Finally an extractor unit was attached to the exhaust for the removal of fumes (Fig. 2).

3.4. Test procedure

In the test, the engine was operated at full throttle. Engine rpm was increased from 2000 rpm to 6000 rpm in intervals of 500 rpm and held for 5 s at each value to allow temperature readings to be taken and then increased steadily over a further 10 s to the next value.

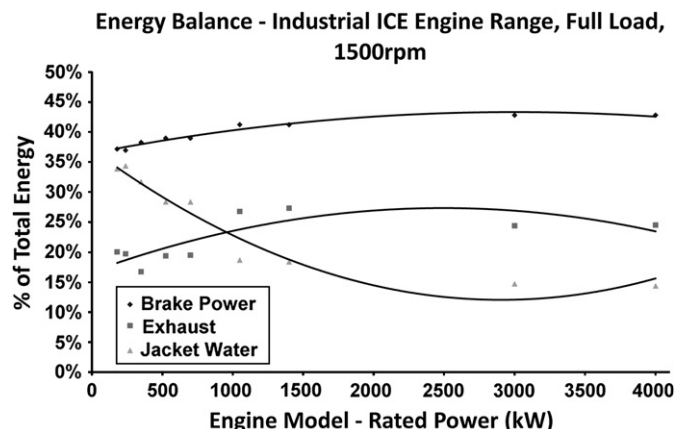


Fig. 1. Industrial ICE recoverable energy balance.

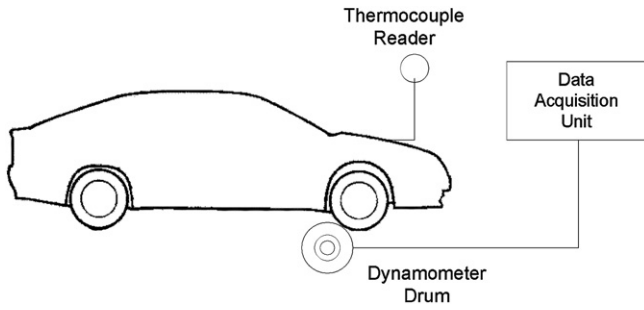


Fig. 2. Dynamometer set-up.

When the final data point has been recorded, the clutch is disengaged and the throttle released. The resultant transmission drag is recorded. Brake power is calculated by the system as the sum of road power and transmission drag.

3.5. Test results

Results for power, torque and exhaust temperature were obtained. Power and torque were plotted against rpm values and are displayed in Fig. 3. The engine was found to produce a maximum brake power output of 87 kW at 6000 rpm and a maximum torque output of 155 Nm at 5000 rpm. The engine oil temperature exceeded 373 K at 4000 rpm, indicating that engine overheating commenced at that speed in this test. An acceptable value for the typical operating speed was judged to be 3000 rpm. The ICE was found to emit 46 kW via the exhaust at this speed, the exhaust gases obtaining a temperature of 812 K. The full temperature regime was between 728 K at 2000 rpm and 1081 K at 6000 rpm. Manufacturer data were not available for comparison.

3.6. Problems encountered

Care had to be taken to ensure that the thermocouple was properly secured in the pipe pocket using the putty. There was a risk of high operating pressures encountered in the exhaust manifold causing the probe to be jettisoned from the pipe explosively. The putty was set 24 h prior to the experiment and allowed to set properly.

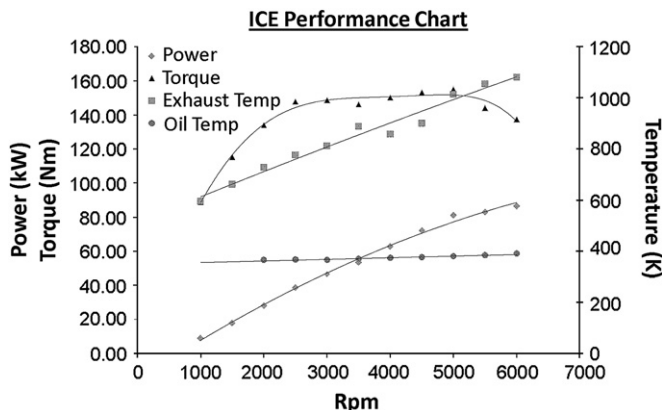


Fig. 3. Performance chart, Renault Safrane 2.0L 16V.

4. Mathematical modelling of Stirling cycle engine

4.1. The Stirling engine – performance modelling

With the information describing the output of the ICE thus available, the problem faced in the feasibility study of the Stirling cycle engine/ICE hybrid automotive engine was the specification of a suitable Stirling cycle engine. The proposal was to model a Stirling engine capable of running on the high grade thermal energy present in the exhaust gas of the ICE. To this end, it was necessary to apply an established mathematical modelling scheme for the design of the thermodynamic gas circuit of the engine. The resulting computations were readily programmed into a spreadsheet and the simulated engine performance displayed by means of indicator diagrams.

4.2. Zero order analysis

The term zero order modelling was first used to describe the basic computations first popularised by Beale [19]. Its genesis lies in empirical observation and experience more than mathematical and scientific principles. As a result, its use is traditionally considered difficult to justify [19,20]. Beale offers an expression for the engine power output in terms of mean cycle pressure, power piston swept volume and operating frequency [21]. Analyses based on the ideal cycle are typically described as zero order as well [19]. Erbay and Yavuz [22,23], Feidt et al [10], Tlili et al [24] among others present alternative models based on the ideal cycle. The equation originally derived by Beale has been expressed in a number of forms. The form adopted for this study was that developed by Walker, West and Senft as indicated by Kongtragool and Wongwises [21]:

$$P = p_m V_p f F \left(\frac{1-\tau}{1+\tau} \right) \quad (2)$$

Eq. (2) offers an expression for the brake power output of the Stirling engine P , as a function of the mean pressure within the engine, p_m , the swept volume of the power piston, V_p the frequency at which it operates, f , and the ratio of the source and sinks temperatures, τ . The term F is an empirical factor. For the ideal cycle, F is 2. A value in the range 0.25–0.35 is offered as being representative of real engines. The equation can be easily manipulated to specify V_p as the subject:

$$V_p = \left(\frac{P}{p_m f F \left(\frac{1-\tau}{1+\tau} \right)} \right) \quad (3)$$

The first step of the model was to approximate the power that might be recovered from the exhaust stream of the Otto engine through a rudimentary energy balance. A typical thermal efficiency for a Stirling engine can be specified as $\approx 30\%$. Therefore, multiplying this by the available energy in the exhaust stream allows an approximate insight into the power output of a Stirling cycle engine running on this thermal source. For the Otto engine under investigation, the thermal output of the exhaust is 46 kW. Therefore, a 30% efficient Stirling engine might convert 13.8 kW of this to recoverable mechanical energy. We proceed with this as the design rating for the Stirling output.

We wish to determine the required swept volume for the Stirling engine to produce 13.8 kW at the nominal 3000 rpm to match the Otto engine speed. We therefore require more information in the form of the source and sink temperatures, the mean engine cycle pressure, and finally we specify the factor F .

The source temperature is 812 K, the temperature of the Otto exhaust, the sink is taken to be 300 K. The mean pressure is arbitrarily specified as 10 MPa and the F value is selected as 0.3.

Under these conditions, Eq. (3) yields a power piston swept volume of 399 cc. This is then used as the initial approximation for the first order analysis.

4.3. First order analysis

The term first order analysis is used in reference to the type of mathematical modelling of the ideal Stirling cycle as first published by Gustav Schmidt in 1871. Although Stirling type engines had been in existence for some half century prior to that, Schmidt, a German engineering professor, was the first to attempt a mathematically rigorous treatment of the thermodynamic cycle involved. The resulting analysis has become the classical analysis of the cycle and remains in use to date [19,25,26]. Schmidt style analysis is highly idealised and the results obtained differ vastly from the actual performance of the engine. This is as a result of the following assumptions made by Schmidt to facilitate the analysis [26]:

- The volume variations in the cylinders are harmonic sinusoidal.
- The gas behaves as an ideal gas ($PV = mRT$).
- The mass of the working fluid is constant i.e. all seals are perfect and there is no leakage from the system.
- The heat exchangers are perfectly effective.
- Cyclic steady state has been established.
- The engine is running at constant speed.
- The kinetic and potential energies of the gas are neglected.
- At any instant the pressure throughout the system is homogenous.

There are numerous treatments of this type available, all closely related to the Schmidt analysis [25–30]. Schmidt-type analysis has found favour due to its mathematical tractability; however various other methods exist. The analysis favoured by the author was a refinement of the Schmidt analysis as advocated by Walker [20]. This analysis provides for design of the thermodynamic circuit using four dimensionless parameters pertaining to the engine geometry; τ , the ratio of the cold end temperature to the hot end temperature T_c/T_h ; κ , the ratio of the swept volumes in the system, $V_{\text{compression}}/V_{\text{expansion}}$; α , the phase angle by which volume variations in the expansion space *lead* those in the compression space; χ , the ratio of the dead volume in the system to the expansion space volume.

The system dead volume is the total volume in the system not swept by one or other of the pistons, such as the volume in the heat exchangers and cylinder clearance volumes.

The use of these dimensionless parameters is advantageous as it permits an analysis independent of the actual geometric considerations, thereby facilitating scaling i.e. any two engines that possess the same numerical values for the above dimensionless parameters possess the same specific work output (work output per unit mass of gas in the system) [25]. Therefore, by analysing in this manner, it is possible to remove geometric design to a secondary phase that can be completed when the necessary thermodynamic design has been completed.

4.4. Results of first order simulation

Using the consolidated design charts presented in Ref. [20], the following results for the dimensionless engine parameters were obtained for a specified $\chi = 0.25$; $\tau = 0.369$; $\kappa = 0.65$ and $\alpha = 1.728$ rad s. Using these ratios in the design equations, the engine parameters as shown in Table 2 were calculated when

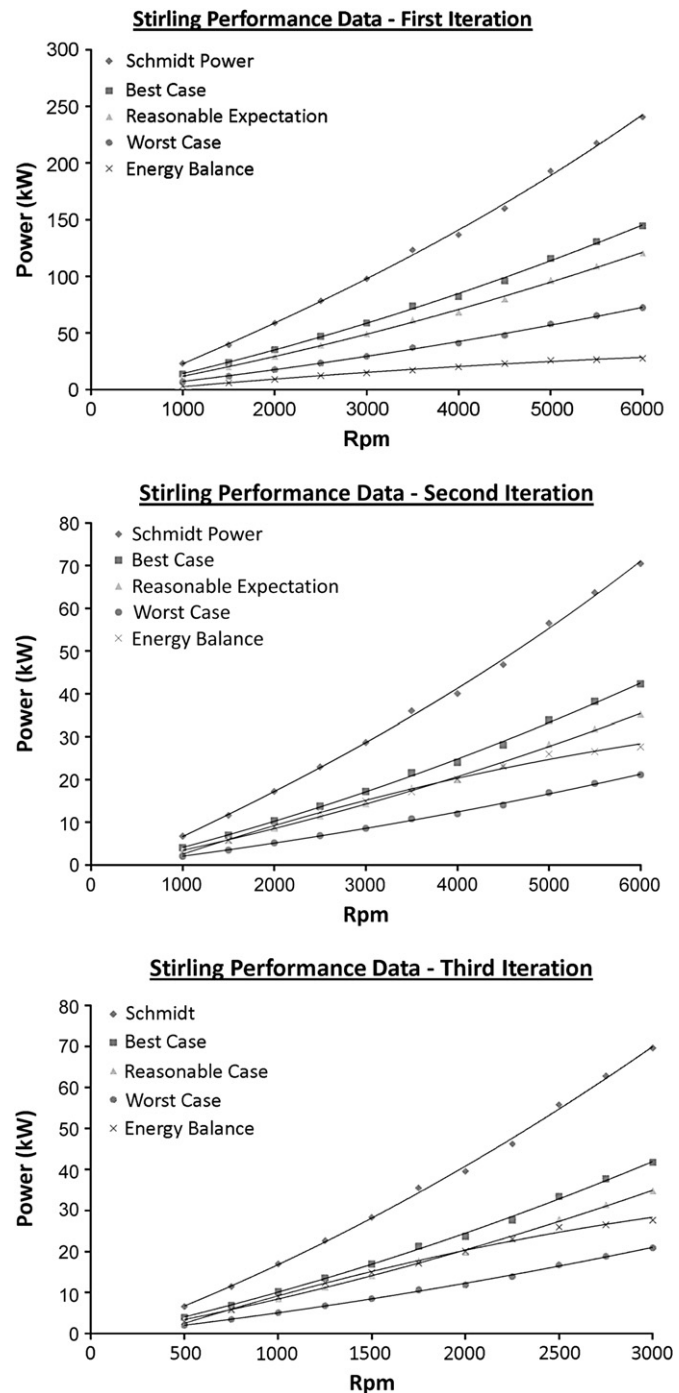


Fig. 4. (a) First order Stirling model – first iteration. (b) First order Stirling model – second iteration. (c) First order Stirling model – third iteration.

a heater temperature of 812 K and cooler temperature of 300 K were utilised:

4.5. Analysis of first order simulation

The above results are obviously extremely optimistic. The thermal efficiency is the Carnot efficiency for the engine, which provides for maximum conversion of heat to work for the given heat source and sink temperatures. An efficiency half that of the Carnot efficiency is considered indicative of a reasonably well designed engine [20]. This would give an efficiency of 31.5% (32%

Table 2

Results of first order simulation – first iteration.

Total working space volume (cc)	659
Brake power output @ 3000 rpm (kW)	48.6
Thermal efficiency, η (%)	63
System mean pressure (MPa)	10

approximately) for the engine. The Curzon–Ahlborn efficiency is of relevance under these circumstances [31,32]. This is a modification of the Carnot efficiency to account for finite heat transfer time between the engine and its source and sink. The expression for efficiency under these conditions is expressed as:

$$\eta = 1 - \sqrt{\frac{T_2}{T_1}} \quad (4)$$

where T_1 and T_2 are the source and sink temperatures respectively. For the given temperatures therefore, the Curzon–Ahlborn efficiency is readily calculated as 39.2% (39% approximately). Curzon and Ahlborn demonstrate that the values computed using this method agree with reasonable accuracy to the best observed performance of real heat engines [31]. However as it offers in this instance a less conservative estimation of the engine efficiency than the previous method, the lower of the two estimates was adopted.

This however leads to the conclusion that the 399 cc compression space engine is oversized; when operating at 32% efficiency, the engine produces 48.6 kW, indicating that it must admit 154 kW at the heated end to provide this power. It is known from the ICE tests that at 3000 rpm, there is an energy availability of 46 kW. This is insufficient and would result in the engine underperforming (“de-rating”). This necessitated a second iteration of the Schmidt analysis to accommodate the energy balance.

4.6. Schmidt analysis second iteration

In order to accommodate the quantity of energy known to be available for transfer at the heated end of the Stirling engine, the value for the thermal energy absorbed in the hot space was specified, allowing the value of the compression space swept volume to be calculated. When the quantity of heat energy required at the heated end was specified as 46 kW, it was found that a compression space swept volume of 76 cc and a consequent total volume of 193 cc were sufficient to provide 14.23 kW at 31.5% efficiency. Results for this analysis are presented in Table 3.

The issue of sizing of the Stirling engine can be seen in Fig. 4a and b. Both figures present models of the simulated Stirling engine performance. On each can be seen lines representing the full Schmidt analysis prediction, the best case prediction, defined as per the literature as being 60% of the Schmidt values, a reasonable expectation prediction, defined as 50% of the Schmidt values and a worst case prediction, defined as 30% of the Schmidt values [20]. Provided also is an energy balance line. This represents 32% of the quantity of energy found to be available in the exhaust stream of the ICE. In Fig. 4a, the Stirling engine is found to be oversized as its closest performance approximation to the energy balance line is the worst case scenario line. This suggests two possibilities; either; (a) the engine operates efficiently (reasonable expectation) but

does not have sufficient quantity of energy available to perform to its full potential, as indicated by the Schmidt mathematical model, thereby underperforming, or; (b) the engine has access to sufficient quantity of energy but is unable to convert it efficiently to work. As it is known from the energy balance that this second case is not possible due to insufficient energy being available from the exhaust stream, it is contended that a smaller engine performing to its reasonable expectation approximation would be able to convert the energy available to useful work more effectively. This is represented in Fig. 4b, where it can be seen that for this size engine the reasonable expectation line more closely approximates the energy balance prediction.

4.7. Schmidt analysis third iteration

The model assumes that the Stirling engine runs at 3000 rpm, the speed of the ICE. There is no way at this level of investigation to predict the engine operating speed under different loads. Engine operating speed is a design issue that involves the power input, the engine torque and an analysis of the gas processes involved. Automotive application particularly involves widely varying load regimes. While it might therefore be feasible to design an engine to operate at the indicated 3000 rpm when the ICE is at that speed, in the absence of a dedicated control system it is unlikely that the Stirling speed would coincide with the ICE speed at every value. It is therefore entirely reasonable to suggest a scenario in which the Stirling engine might run at a speed different to that of the Otto engine. To investigate this, a brief sensitivity analysis of the model was required in which the Stirling engine was assumed to run at a speed different to the Otto engine to which it is attached. In this case, it can be seen that in order to maintain the power output at the given efficiency, an engine of a larger total swept volume is required. This can be seen in Fig. 4c. The figure shows the power curves as before; however in this case the Stirling engine speed is arbitrarily specified as half that of the ICE at each speed of the ICE, while the compression space swept volume has been increased to 150 cc for the compression space in order to provide an approximate match of the reasonable expectation operation scenario to the energy balance curve. The implication of this is that a Stirling engine of a larger total swept volume might be expected to produce the same power at a lower speed for the given power input. This also implies that there exists a possibility to include a variable geometry Stirling engine to manipulate any difference in speed between the ICE and the Stirling engine; however methods of realising this are not investigated here. Variable systems such as stroke control are already an established control method for Stirling engines [19].

5. Results – the combined power plant

Fig. 5 presents the power curve for the Otto engine, the proposed Stirling engine and a compound curve for the proposed hybrid engine. Presented also is a plot of the ratio of Stirling power to Otto power for each rpm value. It is assumed that the Stirling and Otto engines run at the same speeds at each point. It can be ascertained from the graph that the proposed addition of a Stirling cycle engine in this manner could in principle provide approximately an additional 30% shaft power above that provided by the Otto cycle engine alone over the range of typical operating speeds.

The modelling equations used predict a Stirling cycle engine of 193 cc total displacement providing a brake power output of 14.23 kW at the specified 3000 rpm. The engine operates at approximately 32% brake thermal efficiency from a thermal source of 46 kW at a temperature of 812 K.

Table 3

Results of first order simulation – second iteration.

Total working space volume (cc)	193
Brake power output @ 3000 rpm (kW)	14.23
Thermal efficiency, η (%)	31.5
System mean pressure (MPa)	10

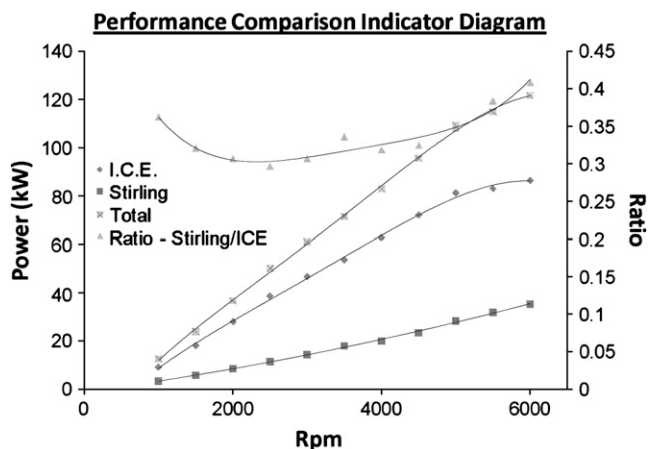


Fig. 5. Power curve for the combined engine.

This addition of a 193 cc total swept volume Stirling engine implies a total volume gain of approximately 10% for the hybrid system above the Otto cycle engine alone (2000 cc).

6. Error analysis

The study presented was completed at a significant level of approximation. The mathematical model of the Stirling cycle engine utilised required the use of a correction factor of 0.5 to align it with realistic operational limits of known Stirling engines. This correction factor accommodates losses in the engine such as heat transfer losses, mechanical efficiencies and fluid losses that are not accounted for in the idealised model but that play significant roles in limiting the actual engine performance. This approach is supported by the literature and is based on experience of actual Stirling engine performance [20]. Although approximated, the model presents important information about the fundamental characteristics of the engine such as working space swept volumes, dead space volumes, source and sink temperatures, pressure and volume variations and power output. More rigorous modelling of the engine system is possible; however this was beyond the scope of this study. Only an approximation of the energy system was required for the feasibility analysis of the system.

7. Conclusions

The study was performed to assess the feasibility of a Stirling cycle/Otto cycle hybrid automotive power plant and to gain some insight into the limits that might be imposed on such a combined engine system.

From the perspective of the energy system, it has been demonstrated that an approximate shaft power gain of 30% is feasible by utilising a Stirling cycle engine as a waste heat recovery device on the exhaust stream of an Otto cycle engine. The corresponding plant swept volume gain is approximately 10%.

However, this study made no attempt to analyse such a set-up in terms of the transient response characteristics, an important aspect of any automotive power plant. Although a rudimentary power gain seems possible, it is likely that the Stirling engine would be at best an awkward match to the Otto cycle engine, as its transient response cannot match that of the latter engine, most likely resulting in considerable lag times. With regard to automotive power plants, response times are generally of considerable

importance; therefore a combined engine, regardless of how efficient it is, is unlikely to be favoured if it cannot offer response times comparable to those of an existing established technology such as the Otto cycle engine. A quantitative analysis of the response times was outside the scope of this study, however; so further analysis or comment is not possible at this stage.

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